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Comparison of Dewar Supports for Space Applications

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Abstract

A simple one dimensional model is developed to illustrate the relative merits of different Dewar support systems. The model considers how different supports effect the parasitic heat load on Dewars in both strength limited and resonant frequency limited applications. The model is used to compare straps, struts, and disconnect supports. The comparison shows that struts are superior in strength limited applications; that straps are superior in resonant frequency limited applications; and disconnect struts are superior when the on-orbit resonant frequency requirement is lower than during launch.

Background

The two cryogenic support systems best suited to long-life space missions are struts and straps. These are illustrated in figure 1. The relative merits of these two types of supports often have been obscured by complex analyses and personal preference of the designers. This paper develops a simple model that highlights the differences of the two systems and the conditions under which each works best.

Space Dewar supports must withstand the loads imposed by all phases of operations (launch, pre-launch ground operations, in space operations, and, in some cases, recovery). The launch loads include not only the combined forces due to gravity and the acceleration of the launch vehicle, but also large dynamic (oscillatory, transitory, and random) loads due to the dynamics of the propulsion system and launch vehicle coupled to the dynamics of the payload. The supports must survive both the peak loads that they transmit to the instrument and the fatigue due to the large dynamic loads. Ground operations can also impose a significant load on the supports. Ground testing, transportation, and launch vehicle integration can require that the system operate in several orientations. Furthermore, ground transportation can put as large peak and fatigue loads as launch. These loads place limits on the ultimate, yield, and fatigue stress in the supports. The on-orbit loads are often significantly lower. If the instrument is recovered and returned to earth, the landing loads can be as large or larger than other loads applied to the system. In particular, large impact loads may occur. In practice, only one of these stresses (yield, ultimate, fatigue) usually limits the performance of a particular design. We will call this limit the "usable" stress, σ_u.

The strength of materials commonly used in supports increases with decreasing temperature. Thus, they are expected to fail at the warm end. This is why the warm end usable stress, σ_u, is used in strength calculations. However, if thermal contractions are not properly accounted for, excessive stresses may occur elsewhere in the support causing failure at unexpected locations. The design of a support system must accommodate these thermal stresses. Otherwise the support will fail with disastrous results.

In addition to these strength limitations, many launch vehicles require the payload to have a high resonant frequency. This places a requirement on the modulus of the supports. The modulus, E, is also a function of temperature. In calculating the stiffness, the mean modulus, $\overline{E} = (1/L)\int_0^L E(T(z))\,dz$, where L is the length, should be used. In practice, E does not vary much with temperature. Thus, we will assume that $\overline{E} \approx E$.

Struts

Struts are made from small diameter thin wall cylindrical tubes with rod end fittings (spherical bearings) at both ends. The rod end fittings ensure that only axial loads are carried by the struts. As a result the struts do not have to take bending loads. One end may also have a length adjustment mechanism. The length adjustment allows the set of struts to be installed and the instrument aligned such that in 0-g the struts are unstressed. In between the ends is the main support tube that is sized for the launch loads. Adhesives are used to attach the tube to the end fittings. The fittings overlap the tubes in a way to prevent stress concentrations in the tube.

Each strut can constrain one mechanical degree of freedom in the supported load. So, a minimum of six struts is needed to fully constrain the load. A typical arrangement is shown in figure 2, where the struts are set in a V formation. The warm ends of a pair of struts are mounted at a common point. The cold end of each strut is mounted at a common point with one from an adjacent pair. These mounting points are placed at 120° around the circumference. The struts are tilted out of the plane of attachment points. Thus the set of struts can constrain both radial, tangential, and axial motion and are compliant to dimensional changes in the system caused by thermal contractions.

Thermal contractions cause the warm and cold attachment points to move with respect to each other. This motion is accommodated by a change in the angle between the struts without stressing the struts. Since the struts operate in both tension and compression, buckling of the struts must be considered in their design. To reduce buckling, the struts are made with filaments wound at an angle to the strut axis. Successive layers have alternative pitches and are sometimes intertwined. The strength, modulus, and thermal conductivity are affected by the winding angle. Thermal radiation in the tube interior can be reduced by filling the strut with insulation.

This system provides the best results and the highest resonant frequencies when the instrument's center of mass lies in the plane of cold attachment points. In a large system, this arrangement of six struts may not provide adequate stiffness to all modes of motion of the load. The mode with the lowest frequency tends to be one in which the plane of attachment points on the load tilts. This mode can be suppressed by placing additional radial struts at some distance from the center of mass. With more than six struts the effects of thermal contractions must be carefully considered. The struts must be placed in positions that not only provide the desired mechanical support but they must also be placed such that all the thermal configurations do not place excessive stress on the system. If thermally induced stresses do occur, they must be considered as part of the total load when analyzing the strength, modulus, and fatigue requirements. The thermal contraction effects vary from operation to operation. The system is assembled at room temperature with some (if any) prestress. On cool down, on the ground, the dimensions of the instrument, the struts and the parts of the insulation attached to the struts will change in dimension. After launch the temperature of the outer attachment points will change. This temperature may vary about some final equilibrium value. During these changes the positions of the outer attachment points, the length of the struts, and attached insulation will change. The stresses in the struts and their attachments must be considered for all three operational conditions as well as for the transitory conditions that occur in between. It is sometimes possible to arrange the struts to virtually eliminate this problem. In the case shown in figure 3, the struts are divided into two sets: one set mounted in a V-formation tilted out of the radial plane that provides the axial, tangential and some of the radial support; and another set, that lies near the radial plane, which only provides radial support. The orientation of the second set of struts is chosen such that any thermal contraction is principally taken up by the strut rotating about its ends.

Straps or tension band supports

Straps are made of two parallel straight sections connected by semicircular end pieces. They have uniform cross section throughout and are made of one or more continuous filaments. The ends are placed on close fitting bobbins. The fit of these bobbins is critical since straps usually fail due to stress concentration in the composite where the strap meets the bobbin. Straps can only carry loads in tension. Thus, they must be placed to oppose each other. One way of doing this is to have two sets of straps. One set placed near each end of the load. A typical arrangement is shown in figure 4, where the straps are placed in a V-formation. This allows axial, tangential and radial loads to be carried. To accommodate the prestress and to constrain all degrees of freedom, more than six straps must be used. Thus the system is inherently over constrained and can lead to difficulties in determining the exact prestress in each strap. The individual straps are not able to carry any appreciable side loads or torques caused by misalignment of their attachments. Thus the bobbins and the rest of the attachment mechanism must have sufficient degrees of freedom or be well enough aligned to ensure that the loads are purely tension and that the loads are placed uniformly across the interface between the bobbins and the straps. A common arrangement is to have two degrees of freedom at each end of the strap. One of these is the axle that holds the bobbin. The other is the pin in the mount on the structure. This pin is set perpendicular to the bobbin.

To ensure that the straps only take tension loads, they are pretensioned. The optimum pretension is set such that under all operational conditions the straps neither exceed their maximum safe design stress nor reach zero applied stress. If a strap becomes unloaded during a period of large oscillating stresses there is the risk that the impact of the reapplied load will damage the strap. Since launch loads are not symmetric and since the center of mass of the load may not be at the center of the supports, there may be more straps at one end than at the other and the pretension may not be the same on all straps. Of course the individual pretensions are not independent of each other. Changing the tension on one strap will effect the tension in all the others. This makes the tensioning and instrument alignment process difficult. The pretension is further effected by thermal contraction effects. As with struts, straps must operate under a number of thermal conditions. The system is assembled warm. On cool down, on the ground, the dimensions of the instrument, the straps, and the parts of the insulation attached to the straps will change in dimension. After launch the temperature of the outer attachment points of the straps will change. This temperature may vary about some final equilibrium value. During these changes the positions of the outer attachment points, the length of the strap and attached insulation will change. The stresses in the straps and their attachments must be considered for all three operational conditions as well as for the transitory conditions that occur in between. A common practice is to delay applying the full pretension until the system is at its final operating temperature on the ground. This pretension will change after launch. However, this may not be a problem since the worst of the stresses (caused by launch) will be over. Thus, it may not matter that the on orbit pretension in the straps change. This change can be minimized by locating the warm attach points so that the thermal contractions of these points result in a rotation of the straps, rather than a change in length.

Comparison of struts vs. straps

It is interesting to compare strut and strap supports. The comparison will be done using a simplified one-dimensional model shown in figure 5. The strut system is modeled as a single strut and the strap system is modeled as a pair of opposed straps. It is assumed that all supports have the same length and that the straps are pre-stressed to 1/2 of their maximum usable stress. We will further assume that the same material is used in both systems. This means that the usable stress, $\square_u$, the modulus, E, and the thermal conductivity, k(T), are the same.

One comparison is to consider the case where the support system design is limited by the maximum stress an individual support can withstand. Consider an axial force, F, that is sufficient to apply the maximum usable stress to the supports. This happens when

$$F = \square_t \square_u A_t \qquad (1a)$$

and

$$F = \square_p \square_u A_p \qquad (1b)$$

where $\square$ is a geometry factor that accounts for the number of supports and their orientation, $\square_u$ is the maximum usable stress, A is the cross-sectional area of the individual supports, and the subscripts (t and p) refer to the strut and strap systems respectively. In the one-dimensional model, the $\square$ are both 1. (In a real system the $\square$ will not, in general, be equal. But, they will depend on the direction of the applied force or torque.) One might expect the amount of prestress to effect equation (1b). But it does not if the prestress is less than F/2 and if the strain is proportional the stress. Under these conditions, as the force is applied, the stress increases in some of the straps and decreases in the opposing straps. Thereby reducing the prestress. When the applied stress is twice the prestress, the opposing strap will become unstressed. The other strap will then carry the full applied stress. As a result of these considerations

$$A_t = A_p. \qquad (2)$$

There are, however, twice as many straps as struts. The total strap area is twice the total strut area. Thus, the straps will conduct twice the heat:

$$2Q_t.= Q_p \qquad (3)$$

This simple model has made a number of assumptions that are not strictly true. The principal one is that the effect of buckling in the strut has not been considered. Buckling causes the strut's $\square_u$ to be less than the strap's. Thus to withstand the same force, A_t>A_p and $2Q_t$>Q_p. In some applications it may be possible to arrange the struts such that the launch loads put the struts in to tension. If this is the case the struts need not be carry a large load in compression. So, the struts need not be strong in buckling. Also, the filaments in the straps are axial while in struts the filaments are at an angle to the axis. This results in $\square_u$ being slightly lower for struts. However straps generally fail at the bobbin at less than the full material strength. At best equation (3) will be approximately true.

Another comparison can be made for the case where the design is limited by a minimum resonant frequency requirement. The comparison will use the same simple model as before. If the mode of interest is the axial mode then the condition that both systems have the same resonant frequency is

$$\omega = \sqrt{\frac{\alpha_t E A_t}{m L}} \qquad (4a)$$

and

$$\omega = \sqrt{\frac{2\alpha_p E A_p}{m L}} \qquad (4b)$$

where m is the mass of the load. The factor of 2 appears in equation (4b) because a displacement, δ, of the load increases the force applied to one strap by $EA_p\delta/L$ while decreasing the force applied to the other strap by an equal amount. So, the force necessary to cause this displacement is $2EA_p\delta/L$. Again, in this simple model the α are both equal to 1. Thus

$$A_t = 2A_p. \qquad (5)$$

Since there are twice as many straps as struts, the total area of the straps is the same as the total area of the strut. Thus,

$$Q_t = Q_p. \qquad (6)$$

The principal questionable assumption in this case is that the α are equal. In a three dimensional system the α depend on the mode of oscillation. If the supports are sized by equation (5), then the mode that is represented by the plane of strut attachment points tilting has a much lower resonant frequency than the same mode in the strap system. If this is the lowest frequency of interest then A_t would need to be increased and $Q_t > Q_p$. However this mode may not be excited and thus not be of concern; or, it may be eliminated as discussed in the preceding section.

<u>Disconnecting supports</u>

The supports discussed above are all compromises between the conflicting requirements of needing high strength during launch and low thermal conductance on orbit. As a result, the supports are sized to meet the launch requirements and one is forced to live with the resulting thermal conductance for the rest of the mission. Supports that disconnect on-orbit can decrease the heat load when the full strength of the support is not needed.

Passively operated devices are one way of achieving this. The passive disconnect makes use of the different environments during launch and on orbit to change passively (and usually reversibly) between high strength and low thermal conductance configurations. The presence (or absence) of large launch forces is used to set the appropriate configuration.

Several different passive disconnects have been proposed and a few have been tested in the laboratory. One design that is likely to be used in space is the Passive Orbital Disconnect Strut

(PODS). These have been developed in several variations. One, PODS III, will be described in detail here.[1] The other PODS variants are modifications that enhance some trait for a particular application. A cut away diagram of PODS III is shown in figure 6.

The warm end is similar to an ordinary strut. It has a rod end fitting and length adjustment. The main support (launch) tube is sized for the launch loads. As in an ordinary strut, there is a second rod end fitting at the cold end. In PODS, the cold end houses the disconnect mechanism. The main tube ends in a housing. Inside there is a second thin wall tube (the orbital tube) that connects the housing to the rod end fitting. Near the rod end there is a lobe that is separated by a small gap from the housing. When an axial force is applied the orbital tube is stretched, or compressed, until the lobe seats on the housing. Larger forces are transmitted by the lobe directly to the housing, bypassing the orbital tube. When the force is removed, the orbital tube relaxes back elastically, re-opening the gap.

This mechanism provides the passive disconnect. On orbit and on ground the gap is open and the load and thermal paths are through the launch and orbital tube in series. The orbital tube is sized for the small ground and orbital loads, providing good thermal isolation. During launch, the load and thermal path is just through the launch tube, thermally and mechanically shorting the orbital tube and providing the required strength. The gap is set to limit the strain in the orbital tube to within its elastic limit. The dimensions and materials for the two tubes are independently designed for their respective application. The launch tube is designed for launch loads and the full temperature span. The orbital tube is sized for the smaller ground and orbital loads and for a restricted temperature span. The disconnect mechanism can be placed at both ends. However, placing one at the warm end is of significantly less benefit. This is a result of the temperature dependence of the thermal conductivity. The housing also provides a good point to anchor thermal radiation shields.

Small side loads can also cause the orbital tube to short against the housing. Such side loads can come from supporting shields and insulation from the launch tube. The design of PODS IV increased the side load stiffness for a slight increase in the orbital thermal conductance by adding a web of radial threads between the rod end and the housing.[2]

<u>Comparison of PODS to straps and struts</u>

In many ways the comparison of PODS to straps is similar to the comparison of struts to straps. So, only the changes caused by the differences between PODS and struts are discussed here. In strength limited applications the PODS will have considerably lower thermal conductance. Because the orbital tube only carries a small load its cross-sectional area is reduced by the ratio of forces involved. For example: if the effective launch load is 10g and the ground load is 1g, then the orbital tube would only need to have 1/10 of the area. In fact the ratio can be even bigger because the orbital tube may not experience as much fatigue as the launch tube. Furthermore, the orbital tube can take advantage of the increased strength and lower conductance of materials at low temperature. Since the orbital tube has a smaller area, it also has a smaller thermal conductance. Continuing with the simple one dimensional model used earlier in this chapter, the heat conducted by the PODS is

$$Q_d = \Box_o(T_c,T_m)\,\frac{A_o}{L_o} \qquad (7)$$

where the subscripts (o, l, c, m, and h) are orbital, launch, cold, mid, and hot respectively, the temperature of the housing, T_m, is determined by

$$\Box_o(T_c,T_m)\,\frac{A_o}{L_o} = \Box_l(T_m,T_h)\,\frac{A_l}{L_l} \qquad (8)$$

and $\Box(T_a,T_b) = \int_{T_a}^{T_b} k\,dT$ is the integrated thermal conductivity. Equation (8) is just a statement of the conservation of energy. Now if the launch tube is the same as the only tube in a strut; the thermal conductance of the two can be compared:

$$Q_d/Q_t = \Box_o(T_c,T_m)\,\frac{A_o}{L_o} \Big/ \Box_l(T_c,T_h)\,\frac{A_l}{L_l} \qquad (9)$$

For the sake of illustration assume the orbital tube is made of the same material as the launch tube and this material has a thermal conductivity of the form $k=\Box T^n$. Then equation (9) becomes

$$\frac{Q_d}{Q_t} \approx \left(\frac{T_m}{T_h}\right)^{n+1} \frac{A_o\,L_l}{L_o\,A_l} \qquad (10)$$

where $(T_c/T_m)^{n+1} \ll 1$ is assumed. Solving equation (8) for T_m and substituting into equation (10) yields

$$\frac{Q_d}{Q_t} \approx \left[1 + \frac{A_l\,L_o}{L_l\,A_o}\right]^{-1} \qquad (11)$$

The improvement in thermal performance depends only on the respective A/L ratios. If the orbital and launch tubes have the same A/L ratio, then $Q_t=2Q_d$. In practice a greater benefit should be realized because the orbital tube can be made of a lower conductivity material. PODS can be compared to straps by comparing equation (3) to equation (11). From this we see that for strength limited applications the PODS will exceed the strap performance .

In the case of resonant frequency limited performance, the comparison depends on how the resonant frequency requirement changes between launch and on orbit. The resonant frequencies are given by

$$\Box_l = \sqrt{\frac{\Box\,\Box_l}{m}} \qquad (12a)$$

and

$$\Box_o = \sqrt{\frac{\Box_o\,\Box_l\,\Box_o}{m(\Box_l+\Box_o)}} \qquad (12b)$$

for the launch and orbital configurations respectively; where $\square$ = EA/L. From this it is seen that $\square_l \geq \square_o$. Combining equations (12a) and (12b) gives

$$\frac{\square_o}{\square_l} = \left[\left(\frac{\square_l}{\square_o}\right)^2 - 1\right]^{-1} \tag{13}$$

If both tubes have the same modulus, E, equation (13) may be substituted into equation (11) yielding:

$$\frac{Q_d}{Q_t} = \left(\frac{\square_o}{\square_l}\right)^2 \tag{14}$$

When there is a significant difference in minimum resonant frequency requirements, PODS offer an improvement in thermal conductance that depends on $(\square_o/\square_l)^2$. A comparison with straps yields a similar result because of equation (6). Thus

$$Q_d < Q_t \approx Q_p \tag{15}$$

If the minimum required resonant frequency is the same on orbit as during launch then PODS is no better than a compound strut or a compound strap. It will have a slightly lower thermal conductance than an ordinary strut because the orbital tube has been optimized for a lower temperature. Thus, for near equal resonant frequencies

$$Q_d \approx Q_t \approx Q_p \tag{16}$$

<u>Summary</u>

It is difficult to extrapolate from this simple model to an actual system as the comparison depends on the details of the application. A detailed analysis for an actual system bears out the results of this simple model.[3]

These comparisons have been based on a rather simple model. For an actual system a more detailed analysis is required. Important features left out of this model include the effects of attaching thermal shields and other masses to the supports. This will effect both the loading on the supports as well as changing the heat flow within the supports. However, this simple model is indicative of the type of results to expect. That in general

1) struts are better than straps in strength limiting cases;

2) straps are better in resonant frequency cases;

3) compound supports offer an improvement, but the increased complexity may only make them worth while in very long life systems or other special situations;

4) disconnect struts offer an improvement over ordinary struts in strength limiting cases;

and 5) disconnect struts offer an improvement over struts and straps in resonant frequency limiting cases when the orbital resonant frequency is significantly less than the resonant frequency during launch.

The key scaling parameters depend on what is limiting the performance. The strength scales as the cross-sectional area, A. The resonant frequency scales as $(A/L)^{1/2}$. In either case the thermal conductance scales as A/L. Since the strength limited case does not depend on L, the thermal conductance can be decreased by lengthening the support. The length can be made as long as wanted until the resonant frequency limit is reached. Thus, if the buckling limit is not reached, A is set by strength requirements and L is set by resonant frequency requirements. Such a support is resonant frequency limited. Thus the optimum choice is straps or disconnect struts. The selection between these two then depends on the change resonant frequency requirements between launch and orbit.

References

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Figures

Figure 1. Diagrams of strut (on left) and strap (on right) supports and their component parts.

Figure 2. Schematic representation of the placement of struts in a Dewar.

Figure 3. An arrangement of struts that eliminates the lowest frequency tilt mode.

Figure 4. Schematic representation of the placement of straps in a Dewar.

Figure 5. One dimensional model of straps and struts.

Figure 6. Diagram of the components of a Passive Orbital Disconnect Strut (PODS).